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On the possibility to improve heat transfer of a sphere by natural convection and water mist

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Abstract. This paper aims to evaluate the effects of water mist on steady natural convection heat transfer from a sphere. Experimental investigations were performed for the Nusselt number (Nu) under a range of water mist concentration in both cases, sphere located in an open volume space with constant surrounding temperature and inside a cylindrical channel with sphere-to-channel diameter ratio ($d/D=0.73$). The results obtained from the related experimental work indicated that the heat transfer rate was enhanced over that for the single-phase air flow as a result of water mist evaporation on the heated surfaces. Compared to air cooling over a range of heat flux investigated, the percentage enhancement in the overall thermal factor was about 40%, 95.4%, 129.4%, and 156.8% respectively for all water mist concentration. In addition, an empirical correlation was also formulated to estimate the real benefits of using the water mist of the enhanced sphere.

1. Introduction

Natural convection heat transfer is an interesting subject that attracted the attention of different researchers not only in the academic research field but in several engineering applications, such as the passive cooling system in nuclear power plants, chemical and electronic cooling devices [1]. Rayleigh number (Ra) and Prandtl number (Pr) considered being the important parameters that influence natural convection heat transfer effectiveness and particularly dominated by the heated surface temperature and surrounding fluid temperature. Based on the heat transfer mechanism, the heat dissipation rate by natural convection is limited at the low-temperature difference and in some cases is not adequate for the operation conditions requirement. Due to the increase in energy demand and cost-saving, the heat transfer enhancement is required more efficient and compact heat dissipation devices with acceptable thermal consuming efficiency. Suspending a fine water droplet (water mist) into air coolant provides an excellent technique of heat transfer enhancement with a significant reduction in size and weight of heat dissipation modules. Using water mist with no higher than 1% of the water phase to dissipate heat offers considerable potential for high-heat-flux thermal management as a result of the direct contact of water mist with the heated surface and subsequent evaporation process [2]. In this study, natural heat transfer enhancement based on the evaporation process of water mist is presented. The effect of deploying water mist on steady-state natural convection flow around a sphere with constant heat flux is experimentally examined and numerically validated.



2. Experimental apparatus and procedures

The experimental apparatus is depicted schematically in figure 1. This apparatus enables to examine the influence of water mist on natural convection heat transfer of the sphere. Experimental measurements were performed with a copper calorimetric sphere as a heated surface under atmospheric pressure conditions, range of water mist concentration and varying surface heat flux. In this study, two cases were investigated: the heated sphere located in an open volume space with constant surrounding temperature ($T=24^{\circ}\text{C}$) and the heated sphere located inside a cylindrical channel with sphere-to-channel diameter ratio ($d/D=0.73$). An electrical heater of 100W is implanted inside the sphere with AC voltage transformer to regulate the input power and achieve the heat flux required. High-thermal conductivity-low-thermal resistance material ($k = 8.5 \text{ W/m}\cdot\text{K}$) was employed to fill the gaps and get better heat conduction between the sphere and electrical heater. The surface temperature of sphere was measured using two calibrated K-type thermocouples placed in positions 1 and 2 as shown in figure 1.

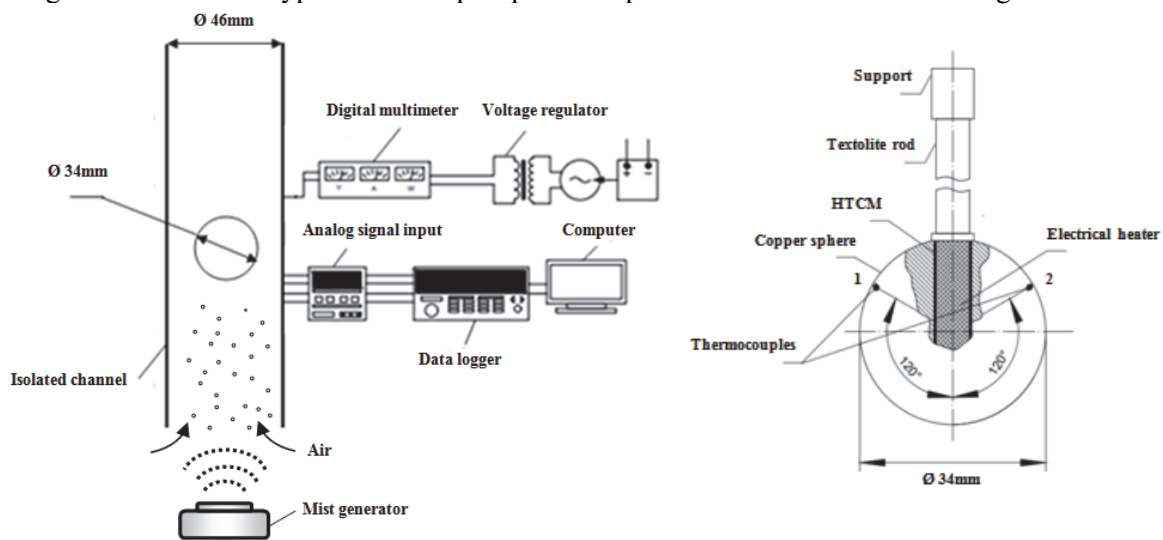


Figure 1. Schematic diagram of the experimental apparatus and the construction heat transfer model.

Micro-sized water mists of 2–10 μm were suspended in the air using an ultrasonic mist generator. The mist generator consists of a piezoelectric transducer whose frequency is 1.7 MHz. This type of mist generator was chosen due to the low power required and a very quiet operation comparing with other mist and evaporative generators. Lang [3] proposed an equation to obtain the average diameter of the water mist droplets based on the capillary wave theory as the formula (1):

$$d_p = 0.34 \left(\frac{8\pi\sigma}{\rho_w F^2} \right)^{1/3} \quad (1)$$

Where σ - surface tension, ρ_w - water density, F - the working frequency of the ultrasonic mist generator. The natural convection heat transfer rate in the terms of Nusselt number (Nu) is calculated according to Eq. (3)

$$h = \frac{q}{\Delta T} = \frac{q}{(T_w - T_\infty)} \quad (2)$$

$$Nu = \frac{hd_{sp.}}{k} \quad (3)$$

Where d_{sp} - sphere diameter, k - thermal conductivity. Based on the temperature difference (ΔT) and sphere diameter (d_{sp}), the Rayleigh number (Ra) can be determined as follow:

$$Ra = \frac{\Delta T \cdot \rho^2 \cdot g \cdot \beta \cdot d_{sp}^3}{\mu^2} \cdot Pr; \beta = -\frac{1}{\rho_{\infty}} \left(\frac{\partial \rho}{\partial T} \right)_p \quad (4)$$

The Weber number is defined as (5):

$$We = \frac{1}{N} \sum_i^N \frac{j^2 d_p}{2 \rho_w \sigma} \quad (5)$$

Where j - water flux density and d_p - water mist diameter. According to Moffat [4], the uncertainty analyses of the experimental results are performed by defining the error of the measured parameters and estimating its effect on the dependent variables. The maximum uncertainties in parameters h , Ra are 5 % and 5.27%, respectively.

3. Physical model and computational methods

A copper sphere of 34-mm diameter under a range of surface heat flux was defined in a cylindrical domain and an open volume space domain with atmospheric pressure conditions. The computational domain involves the fluid entering section, sphere section, and outlet section. The diameter of the cylindrical channel is 46 mm and sphere-to-channel diameter ratio d/D is considered to be 0.73. The computational modeling includes the prediction of hydrodynamics and heat transfer behaviors using air as the working fluid. Some boundary conditions were needed for CFD simulation such as the surface heat flux, inlet air temperature and channel wall condition. Based on the temperature difference and sphere diameter, six Rayleigh numbers, changing from 0.9 to $1.9 \cdot 10^5$, and inlet surface heat fluxes, varying from 0.27 to 1.3 kW/m², are selected as the boundary conditions in the present simulation study. The computational modeling was performed using the commercial software SW flow simulation. The steady Reynolds-averaged Navier-Stokes (RANS) equations were adopted in the terms of mass, momentum and energy conservation in Cartesian coordinate system as follow: [5]

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial t} (\rho u_i) = 0 \quad (6)$$

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i u_j) + \frac{\partial p}{\partial x_i} = \frac{\partial}{\partial x_j} (\tau_{ij} + \tau_{ij}^R) + S_i, i = 1, 2, 3 \quad (7)$$

$$\frac{\partial \rho H}{\partial t} + \frac{\partial \rho u_i H}{\partial x_i} = \frac{\partial}{\partial x_i} (u_j (\tau_{ij} + \tau_{ij}^R) + q_i) + \frac{\partial p}{\partial t} - \tau_{ij}^R \frac{\partial u_i}{\partial x_i} + \rho \epsilon + S_i u_i + Q_H, \quad (8)$$

The SST $k-\omega$ turbulence model was employed in this simulation to obtain an optimal model formulation that accurately predicts the turbulent effect near the surface and improve the surface shear stress and heat transfer predictions.

4. Results and discussion

4.1. Validation of heat transfer from single sphere to fluid

The present experimental results of the single phase - air coolant are reported as the references data. In order to assess the reliability of the measurements and experimental apparatus, the heat transfer behaviors in the terms of Nu with Ra number were validated with the correlations of Churchill and Chu [6], and Lefebvre [7] as shown in figure 2. As found, the deviations of the average Nu numbers are within $\pm 1.5\%$ of those from Churchill correlation while the present Nu numbers are within $\pm 4.8\%$ of those obtained from Lefebvre correlation.

Churchill correlation [6]

$$Nu = 2 + \frac{0.589Ra^{1/4}}{[1 + (0.469 / Pr)^{9/16}]^{4/9}} \quad (9)$$

Lefebvre correlation [7]

$$Nu = 0.49Re^{0.25} \quad (10)$$

In addition, the experimental results for both cases were compared with those obtained from the numerical simulation. The deviations of the experimental Nu numbers of sphere were within 16% and 20% for the open volume space and inside cylindrical channel domains, respectively. Figure 3 depicts the numerical simulation results in the form of a velocity field and vectors indicating the flow direction.

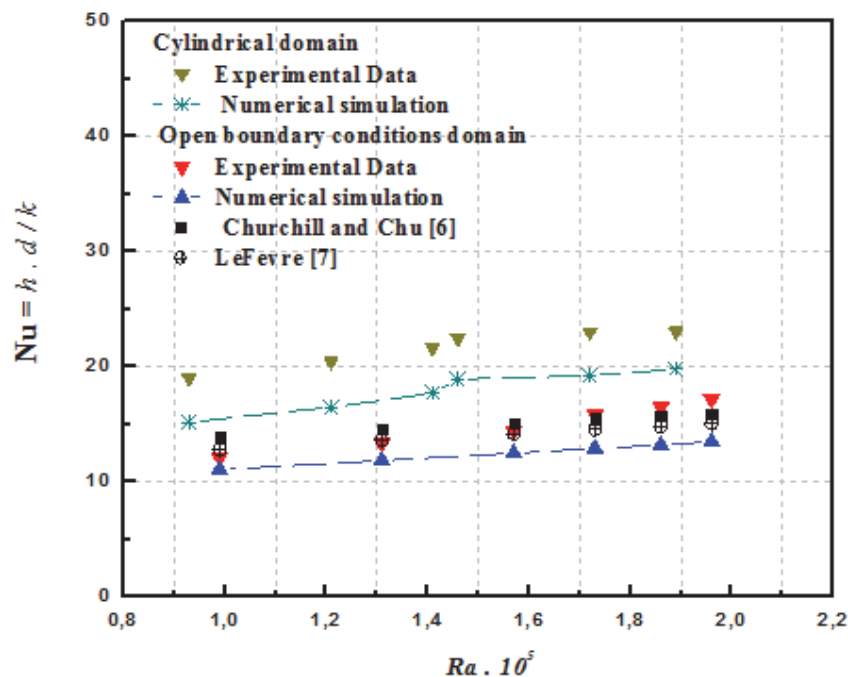


Figure 2. Comparisons between the average Nusselt number with Rayleigh number of the sphere for the present work and those obtained by previous correlations.

4.2. Analysis of heat transfer with water mist

Several heat fluxes were conducted to evaluate the effect of surface temperature ($50^{\circ}\text{C} - 115^{\circ}\text{C}$) on natural heat transfer characteristic of sphere. Figures 3 and 4 show the variation of average Nu number with Ra number for both cases under a range of water mist concentration. It can be seen that the Nu number in single-phase air cooling tends to increase gradually with increasing Ra number. By introducing the water mist, the effect of water mist evaporation appear and the average surface

temperature has dropped dramatically from 115°C to 61.3°C under ($j=111.68 \text{ kg m}^{-2} \text{ hr}^{-1}$). It can be seen that the Nu number increases as water mist increases for constant heat flux. The heat transfer enhancement by water mist relative to the single-phase air cooling was more considerable at higher water mist concentration. The Nu number values by water mist were respectively 39.4%, 95%, 128.7% and 155% higher than those obtained by air cooling for the range of water mist of ($j = 23.39 - 111.68 \text{ kg m}^{-2} \text{ hr}^{-1}$).

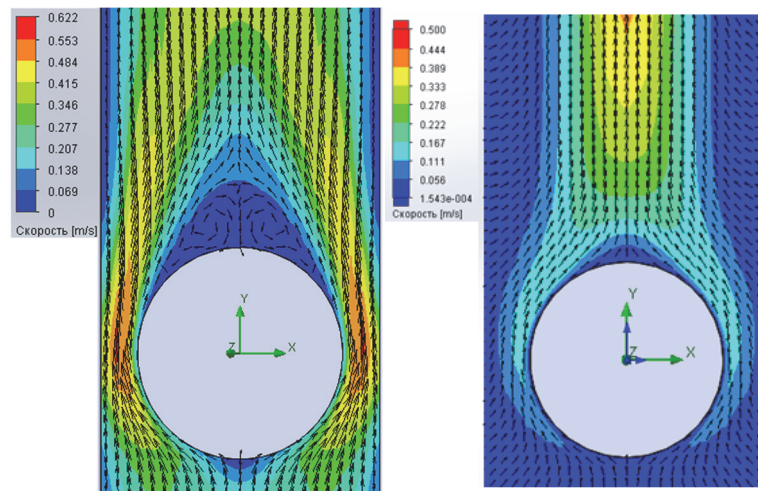


Figure 3. Flow visualization; a – inside cylindrical channel domain; b – open volume space domain.

For sphere located inside a cylindrical channel, the enhancement of heat transfer is different from that of the open volume space due to the effect of cylindrical channel boundary conditions. It can be seen that the Nu number increase significantly as Ra number increase then starts decreasing. The Nu number values were 36.5%, 90%, 117% and 145% higher than those obtained by air cooling for water mist range considered. The heat transfer mechanism of heat transfer enhancement by water mist may involve three important different parts: water mist evaporation on the sphere surface, convective heat transfer based on a decline temperature gradient due to water mist evaporation in the air flow near the sphere surface and enthalpy transport based on water mist departure which is effectively used for cooling. In addition, the heat transfer enhancement depends on the specific heat (C_p) of the mixture affected by the fluid temperature and its value decreases while the temperature drops from the T_1 (air temperature) to T_2 (mixture temperature). The ratio of the Nu number for the sphere exposed to water mist (Nu_{mist}) to that for the sphere when water mist is not suspended (Nu_0) at a similar heat flux can be defined as the thermal enhancement factor. Figure 7 depicts the thermal enhancement factor with Ra number under the range of water mist concentration.

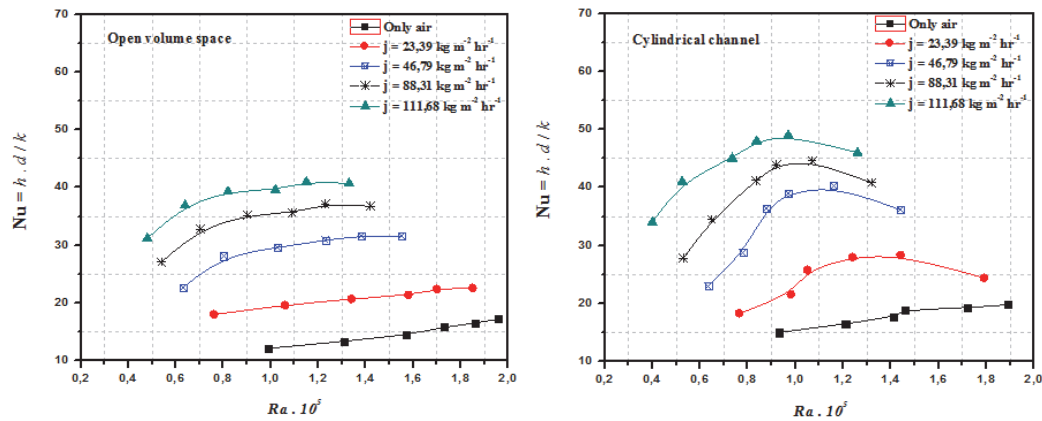


Figure 4. Variations of Nusselt number with Rayleigh number for various water mist concentration.

Evidently, the thermal enhancement factor for all cases increase with increasing water mist concentration. This reflects that the enhancement factor becomes more significant when using water mist at low or high concentration. The average thermal enhancement factor is 40%, 95.4%, 129.4% and 156.8% higher than those in air-cooling for all cases of water mist, respectively. For the sphere located inside a cylindrical channel, the enhancement factor increase as Ra number increase, then then starts decreasing due to the surface wetting and liquid film formation. The experimental results of the enhancement factor were developed using the method of least square regression analysis, in which a dimensionless Rayleigh number (Ra) and Weber number (We) were taken into account. The resultant empirical correlation is shown below in equation (10) with absolute deviations between the experimental and predictions data of $\pm 5.6\%$.

$$\frac{Nu_m}{Nu_0} = 1 + Ra^{0.34} We^{0.5} \quad (10)$$

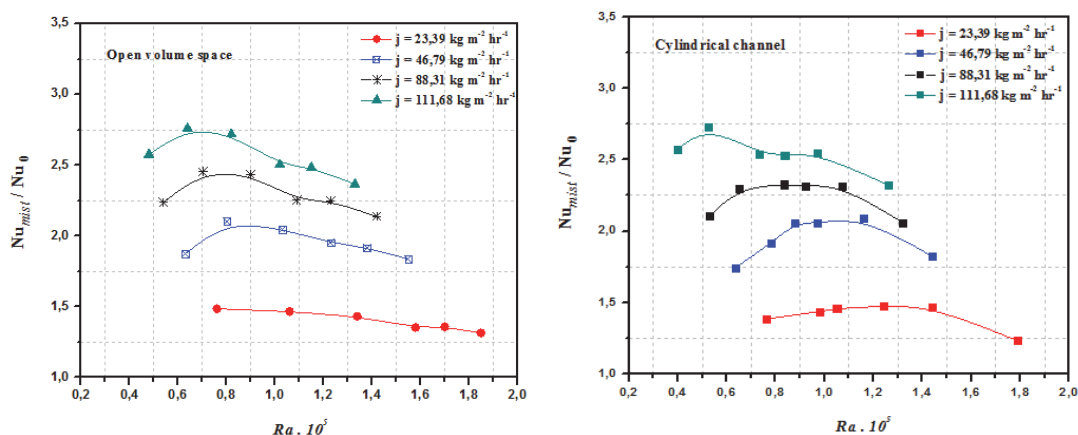


Figure 5. Variations of thermal enhancement factor with Ra number for various water mist concentration.

Conclusions

Natural heat transfer enhancement based on the evaporation process of water mist is presented. The main observations can be summarized as follows: suspending water mist in air cooling is a very efficient way to remove a great deal of heat from the sphere surface. The heat transfer rate in the terms of Nu number increases with increasing water mist concentration. The Nu number increases about 39.4%,

95%, 128.7% and 155% as compared with air - cooling by suspending different water mist concentration. From the experimental results, an empirical correlation for the thermal enhancement factor has been developed against the Ra number and We number.

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